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DEVELOPMENT OF A VERSATILE SYSTEM FOR DETAILED STUDIES ON THE PERFORMANCE OF HEAT PIPES

by

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Technical Report No. ME-TR-64

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ABSTRACT

A heat pipe with variable dimensions was designed for the study of steady state and transient heat pipe performance using different fluids and wicking materials.

An open ended dewar was designed and constructed for housing the heat pipe system. The maximum length of wicking material was 82 cm; this distance was considered the maximum length of heat transfer required in future space suits. Distilled water was the transfer medium used in the wicking chamber.

The heat input to the dewar was supplied by electric heaters. Circulation of cool water was used to remove heat from the condenser end of the dewar. Approximately 45 thermocouple points were used for measuring important temperatures in the system.

The maximum heat transfer capability or wick "burn out" point, was 10 watts with a wick length of 81.9 cm and operating temperatures of 26.7°C (80°F) ±5°C. For the Refrasil #C100-28 used, the 10 watt "burn out" point corresponds to 0.594 watts per cm width of the wick.

The required transport rate per cm^2 area at "burn out" (10 watts) was 0.361 $cm^3/min\text{-}cm^2$. This value was well within the 0.299 to 0.424 $cm^3/min\text{-}cm^2$ range predicted by horizontal wicking tests performed by the author on the Refrasil #C100-28.

Throughout the entire wicking chamber, a maximum temperature variation of ±½°C. was encountered during normal heat pipe operation. No transient temperature lag from one end of the wicking chamber to the other end was observed during heat input changes. Apparently the time constants of the heat input changes were much larger than the temperature equalizing time constant of the wicking chamber.

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1. INTRODUCTION

During future space exploration, astronauts will remain in outer space for extended periods of time. Extravehicular tasks will be performed by the astronauts and their only protection from the surrounding space will be a space suit.

A thermoregulatory system will, by necessity, be incorporated in the space suit. Its purpose will be to control the temperature of the atmosphere immediately surrounding the astronaut (the space between the astronaut's body and the inner layer of the space suit) and, in general, to provide a satisfactory thermal environment for him.

At present, liquid cooled undergarments are used for the temperature regulation. An intricate system of valves, pumps, and auxiliary equipment is required. Need for a more efficient, less complicated, and self-contained thermoregulatory system seems apparent.

A new system must be capable of rejecting heat from the space suit and of transferring heat from one part of the suit to another part. The heat rejection could be either by radiation to outer space or by heat exchange with a porous plate sublimator.

A heat pipe system for transferring heat has many advantages over the water-cooled undergarment. First, the heat pipe is a completely closed system and needs no recharging after initial assembly. Heat conducting capabilities of 100 to 1000 times that of the best conducting metals can be obtained using heat pipes. There exists no need for pumps, compressors, or auxiliary equipment during the operation of the heat pipe. If a temperature difference exists between the ends of the heat pipe, heat will be transferred down (evaporator to condenser) the pipe.

Regardless of the heat pipe orientation, relative to the gravitational field, the pipe will transfer heat in either direction as long as a wicking assembly is present to return the liquid to the heated end. The heat pipe is particularly suitable for "hard" suits now under consideration for future space exploration.

2. HEAT PIPE OPERATION

There are three distinct portions or sections to a heat pipe;

1. a heat input (evaporator) section,
2. a mass (vapor state) transfer and a mass (liquid state) return section, and
3. a heat rejection (condenser) section.

The heat input evaporates the liquid to the transfer section. As a result of a very small pressure gradient, the vapor is forced down the pipe to the heat rejection section where the vapor is condensed. Either by capillary action or gravitational feed, the liquid is then carried back to the heat input section for recycling.

When capillary action is required for returning the liquid to the heat input section, some type of wicking material must be used. The wicking materials normally used are: fine wire screens, porous solid materials, and natural or synthetic cloths.

3. GENERAL THEORY

The heat pipe considered in this paper is shown in Fig. 1. The double open-ended dewar is assumed to be a perfect insulator in the radial direction. All heat added at the heat input section is transferred axially down the dewar. The major design consideration for the setup was to provide means for the evaluation of wick performance with a well defined transfer length within the heat pipe.

A temperature gradient of less than ½°C existed over the entire length of the mass transfer section. This nearly constant temperature existed because of the very small pressure gradient down the length of the transfer section.

The power transfer capability of a heat pipe depends on the following basic parameters:

i. the capillary pumping head ΔP_c,

ii. the vapor pressure drop ΔP_v,

iii. the liquid viscous drag ΔP_L, and

iv. the gravity head ΔP_g [1]*.

There also is a pressure drop term caused by a momentum change in the flow of the returning liquid. This term, however, is negligible when compared to the other four terms [2]. From a pressure balance standpoint only, the equation for heat pipe operation is

$$\Delta P_c \geq \Delta P_v + \Delta P_L + \Delta P_g \ . \tag{1}$$

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*Numbers in brackets designate references.

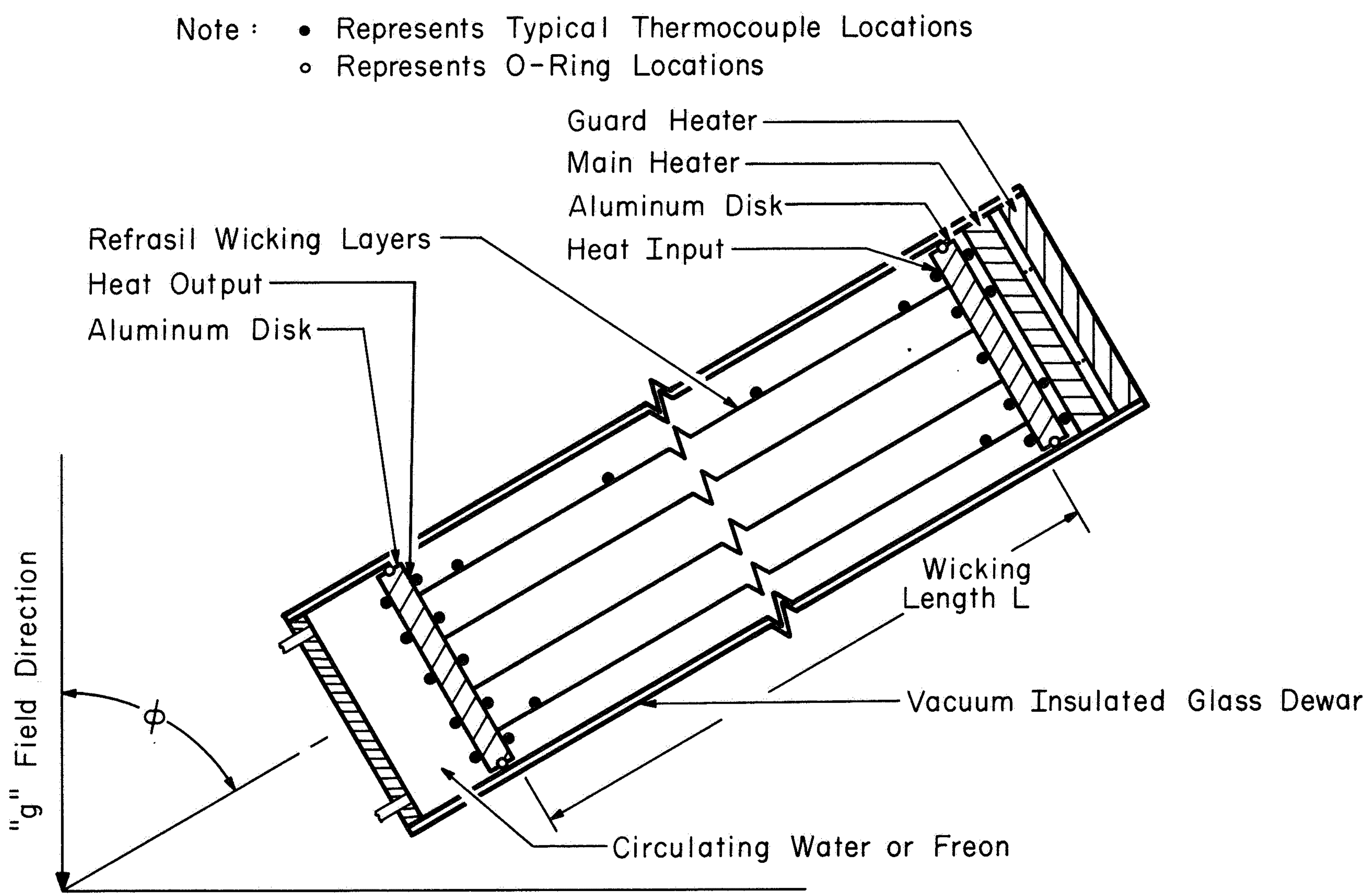

Figure 1. General Heat Pipe Design

In other words, the available pumping head must be sufficient to overcome the pressure losses caused by vapor transport, and by the viscous drag of the returning liquid. Gravity can aid or hinder, depending on the orientation of the evaporator with respect to the condenser. The following theory is patterned, in part, after Feldman's [3] analysis of heat pipe operation.

3.1 CAPILLARY PUMPING IN WICK

The capillary pumping term, ΔP_c, may be written

$$\Delta P_c = 2\sigma \left(\frac{\cos \theta_e}{r_e} - \frac{\cos \theta_c}{r_c} \right) , \qquad (2)$$

where σ is the surface tension of the liquid, θ_e is the liquid contact angle at the evaporator section, θ_c is the liquid contact angle at the condenser section, r_e is the effective radius of the wick pore at the evaporator section, and r_c is the effective radius of the wick pore at the condenser section.

The maximum value of Eq. (2) is obtained when $\theta_e = 0°$, $\theta_c = 90°$, and r_e is the actual pore radius of the wicking material. Equation (2) becomes

$$\Delta P_{c\ max} = 2\sigma \left(\frac{1}{r} \right) = \frac{2\sigma}{r} , \qquad (3)$$

where $r = r_e$.

3.2 VISCOUS LOSSES IN VAPOR

Pressure losses caused by vapor flow in the heat pipe may be determined by using existing theory for either laminar or turbulent flow in pipes [4]. When the vapor flow pressure drop was calculated, it was found to be negligible when compared to other pressure losses in this heat pipe.

3.3 VISCOUS LOSSES IN LIQUID

Flow rates and velocities encountered with capillary flow in wicks may be assumed to be laminar* and relatively free from inertial effects. Darcy's Law [5] for flow through porous media then can be applied to yield

$$\Delta P_L = \frac{\mu L \dot{m}}{\rho K A} = \frac{\mu L \dot{v}}{K A} , \qquad (4)$$

where μ is the liquid viscosity, L is the wicking material length, $\dot{m}$ is the liquid mass flow rate, ρ is the liquid density, K is the wick permeability, A is the total cross sectional area of the wick, and $\dot{v}$ is the volume flow rate.

3.4 GRAVITATIONAL FIELD EFFECTS

The gravitational field can aid, hinder, or have no effect on the liquid flow in the wick. This effect depends upon the orientation of the evaporator and condenser sections relative to the direction of the gravitational field. The general equation for the pressure loss in the wick caused by gravity is

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*For the heat pipe considered in this report, the Reynolds numbers were 56.3 and 14.5 for the water flow in the Refrasil and the vapor flow in the wicking chamber, respectively.

$$\Delta P_g = \rho g L \cos \phi \ , \tag{5}$$

where ρ is the liquid density, L is the wick length, g is the acceleration of gravity, and ϕ is the angle between the heat pipe axis and the gravitational field as shown in Fig. 1. The algebraic sign in Eq. (5) is

(+) when the evaporator is above the condenser in the gravitational field (hinders),

(-) when the condenser is above the evaporator in the gravitational field (aids),

and when the evaporator and the condenser are in a horizontal plane gravity has no effect, the term is zero.

The gravitational effects on the vapor flow are neglected in this report, since the density of the vapor is approximately 4000 times less than that of the liquid at 26°C.

3.5 MAXIMUM FLOW RATE IN THE WICK

By substituting Eqs. (3), (4), and (5) into Eq. (1), the maximum flow rate is obtained as

$$\dot{m} = \frac{\rho K A}{\mu L} \left(\frac{2\sigma}{r} - \rho g L \cos \phi \right) . \tag{6}$$

If the heat pipe is horizontal, Eq. (6) reduces to

$$\dot{m} = \frac{2\sigma\rho K A}{\mu r L} = \dot{v}\rho \tag{7}$$

since $\phi = 90°$.

For given operating conditions, wicking fluid, and wicking material σ, ρ, μ, K, A, and r are constant and Eq. (7) becomes

$$\dot{m}L = \frac{2\sigma\rho KA}{\mu r} = \dot{v}\rho L = C = \text{constant}. \tag{8}$$

3.6 MAXIMUM HEAT TRANSFER RATE

Because the rate of heat transfer attributed to latent heat transport is large and the temperature gradient along the heat pipe is small, conduction, radiation, and sensible convection heat transfer will be negligibly small. Therefore, assuming all thermal energy is transferred as latent heat, the heat transfer rate is

$$Q = \dot{m}h_{fg} = \dot{v}\rho h_{fg} \quad , \tag{9}$$

where h_{fg} is the latent heat of vaporization at the operating pressure and temperature of the system.

Combining Eqs. (6) and (9), the maximum heat transfer rate becomes

$$Q = \frac{\rho KA}{\mu L}\left(\frac{2\sigma}{r} - \rho gL \cos\phi\right) h_{fg} \ . \tag{10}$$

If the heat pipe is horizontal, Eq. (10) reduces to

$$Q = \left(\frac{2\sigma\rho KA}{\mu rL}\right) h_{fg} \tag{11}$$

since $\phi = 90°$.

The following assumptions were made in obtaining Eqs. (10) and (11):

1. Gravity effects on vapor flow are negligible.
2. Liquid flow in wick capillaries is laminar.
3. Viscous vapor flow losses are negligible.
4. Conduction, radiation, and sensible convection heat transfer along the heat pipe are negligible.
5. Liquid properties are constant along the heat pipe.
6. Flow and heat transfer are essentially one-dimensional.
7. Wick is uniform and evenly saturated.
8. Heat transfer is uniform over the evaporator and condenser surfaces.

4. EXPERIMENTAL WORK

4.1 HEAT PIPE DESIGN*

Dewar

A 7.62 cm inside diameter, 8.90 cm outside diameter, and 101.5 cm long double open-ended, silvered glass dewar was used to contain the entire test setup. The inside and outside diameters were attained by using concentric glass cylinders with a high vacuum between them and sealed on the ends. This vacuum, between the silvered surfaces, provided the necessary insulation between the heat transfer area and the surrounding atmosphere†. The three main elements of the heat pipe previously mentioned were then fitted inside this dewar with a combination of o-rings and gaskets sealing axially down the dewar. A radial connection between the inside of the dewar and the exterior was provided. This connection was needed to allow charging the wicking chamber with the desired transfer medium or fluid and to provide access to the wicking chamber for thermocouple wires.

Heat Input Section (Evaporator)

The main requirement here was to have a heat input which could be measured accurately. The final system chosen was to use two electrical heaters. The main heater, closer to the heat transfer area, was used for the total heat input and the distant guard heater was used to

*Initial design of the heat pipe was by Prof. John C. Chato of the University of Illinois, Urbana, Illinois.

†Unsilvered window strips allowed inspection of the interior after assembly.

minimize heat flux in the outward axial direction. By measuring the input wattage to the main heater, an accurate measurement of heat input to the system could be obtained. Both heaters were supplied electrical energy from variable transformers and the wattage inputs were measured by wattmeters (see Fig. 2).

Heat Transfer Section (Wicking Chamber)

The wicking material used was Refrasil #C100-28. This material had a water lift rate and horizontal transfer rate* which were deemed the best from past experience.

Rather than arranging the wicking material in a cylindrical fashion, concentric with the dewar as is normally done, it was decided to assemble the wicking material in four horizontal layers. The two center strips were 5.0 cm wide while the top and bottom strips were approximately 3.8 cm wide. This arrangement eliminated any gravitational effects on a given cross section while the dewar was maintained in a horizontal attitude. Also a wicking cage was utilized for suspending the Refrasil (see Fig. 3). This cage consisted of four teflon disks, two stainless steel rods, and three intermediate supports. The three intermediate supports were required because the wicking cage was the primary device separating the two aluminum disks. Since a vacuum of approximately 10^{-3} mm of Hg. was pulled in the chamber between the disks, the wicking cage was required to support a load of approximately 100 pounds in compression. Finally, circular sections of Refrasil were attached to the insides of the aluminum disks for the purpose of

*See p. 27 for a plot of Transfer Distance vs. Time for the Refrasil #C100-28 and other Refrasils commonly used.

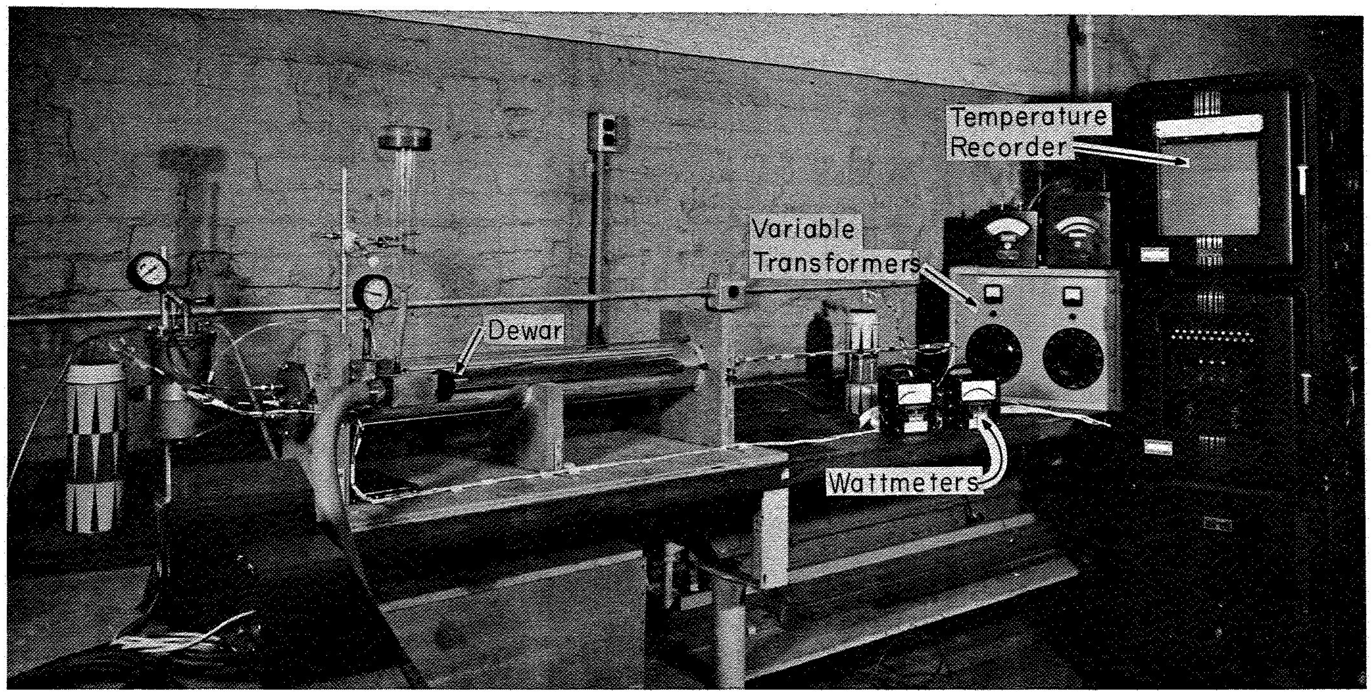

Figure 2. Heat Pipe System

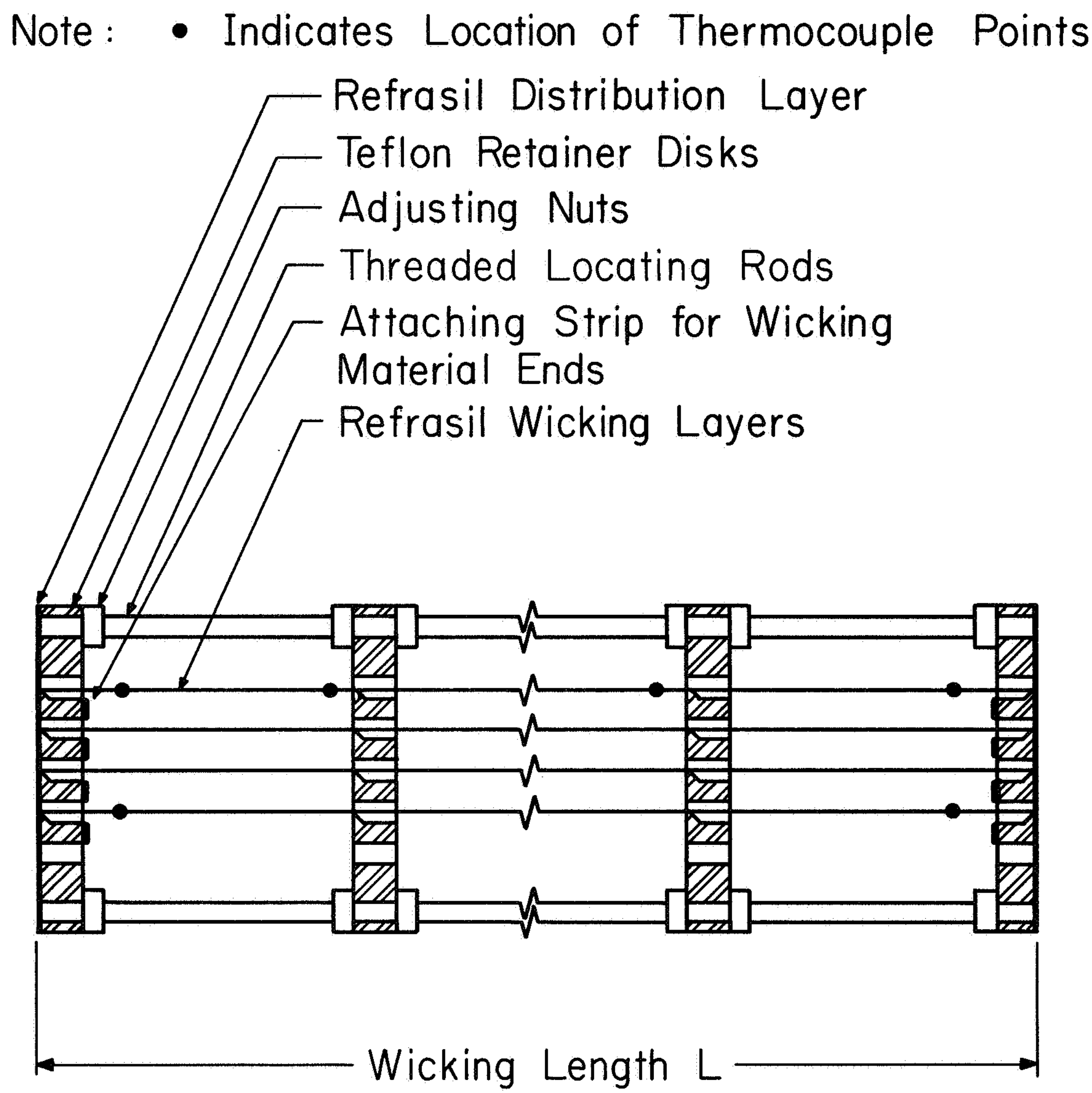

Figure 3. Wicking Cage Assembly

equalizing the distribution of fluid on each disk. These sections of Refrasil were lightly spot-epoxied to the aluminum disks.

Heat Removal Section (Condenser)

The heat transferred down the dewar by the wicking chamber was dissipated through the condenser aluminum disk. After the heat was conducted through the condenser aluminum disk, it was removed by circulating cool water through the condenser chamber and expending the water. An alternate method, which will be used in the future, uses a Freon in a liquid-vapor state circulating in a closed system. The closed system consists of the condenser end of the dewar and a storage tank (see Fig. 4). Copper cooling coils are located inside the storage tank and will be used to remove the heat from the Freon so that the pressure of the Freon in the tank remains within a reasonable range and that the temperature can be controlled to remove all the heat transferred. Water or any other cooling agent can be circulated through the cooling coils since the coils are isolated completely from the Freon inside the tank.

Temperature Measurement

Temperatures throughout the system were measured by copper-constantan thermocouple wires. Approximately forty-five different points in the system were monitored. Thermocouples were placed across mica disks between the two heaters for measuring the gradient existing there. Next, thermocouples were placed across the heater aluminum disk in order to determine the temperature gradient for heat input calculations. In the wicking chamber, six thermocouple points were distributed, two at each end and two distributed down the center of the wick for

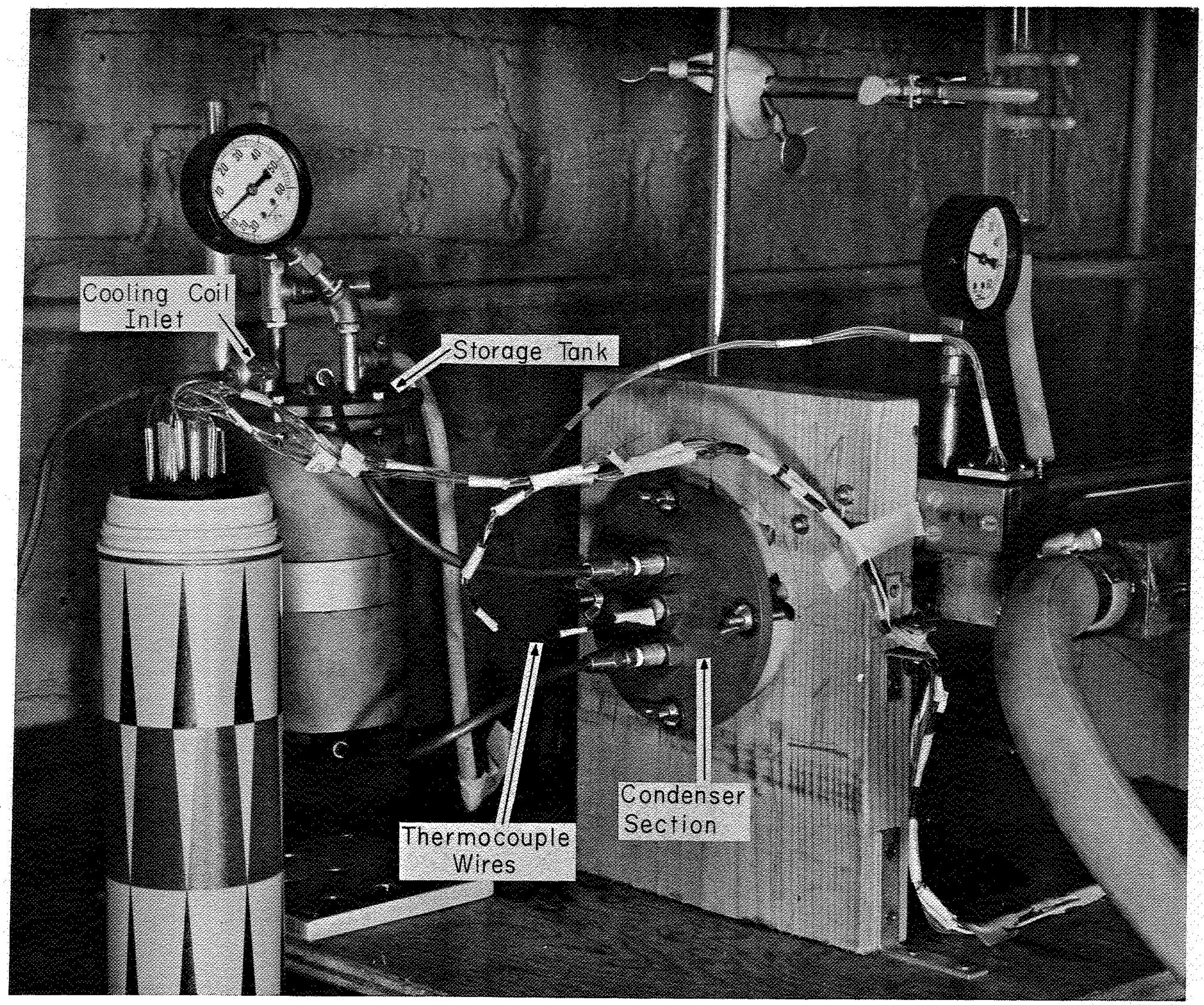

Figure 4. Heat Removal System

measuring temperatures and for determining when "burn out" (desiccation of the wick material) occurred. At the condenser end, thermocouples were used again to determine the temperature gradient for determining the heat flux out of the system. Accurate determination of the conductivity of the aluminum disks was made before the system was assembled. The product of conductivity and temperature difference indicated the heat flux.

Wicking Chamber Manifold

A manifold was constructed for use with the wicking chamber. First, the manifold had a large vacuum valve (see Fig. 5) which connected to a vacuum system (see Fig. 6) capable of vacuums to 10^{-6} mm of Hg. for evacuating the wicking chamber. A transfer plug was adapted to the manifold for extracting the thermocouple wires from the wicking chamber. A small needle valve was attached to the manifold for charging the chamber with the working fluid. A union was also adapted to the manifold for connecting a combination pressure-vacuum gage for indicating the pressure in the chamber. Another union was adapted to the manifold for making the direct connection to the wicking chamber. The entire manifold, along with the valves and unions, was mounted to the dewar mounting stand near the wicking chamber outlet tube (see Fig. 7). Then, all of the necessary connections to the wicking chamber were made through the wicking chamber manifold.

Temperature Recording

A twenty-one point Leeds and Northrup millivolt recorder was used (see Fig. 2). Since all forty-five thermocouple points could not be recorded, the most representative points were chosen after monitoring

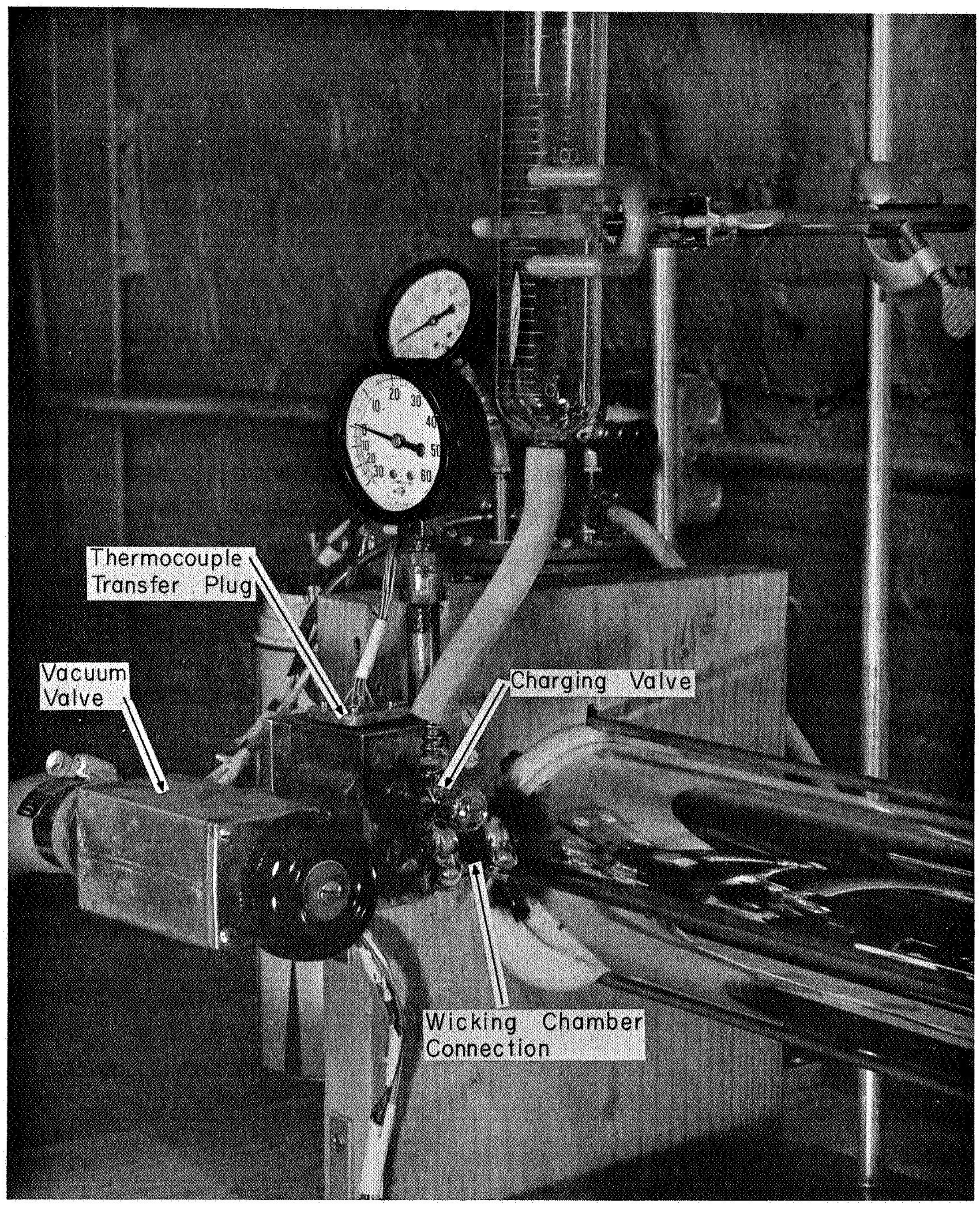

Figure 5. Wicki-g Chamber Manifold

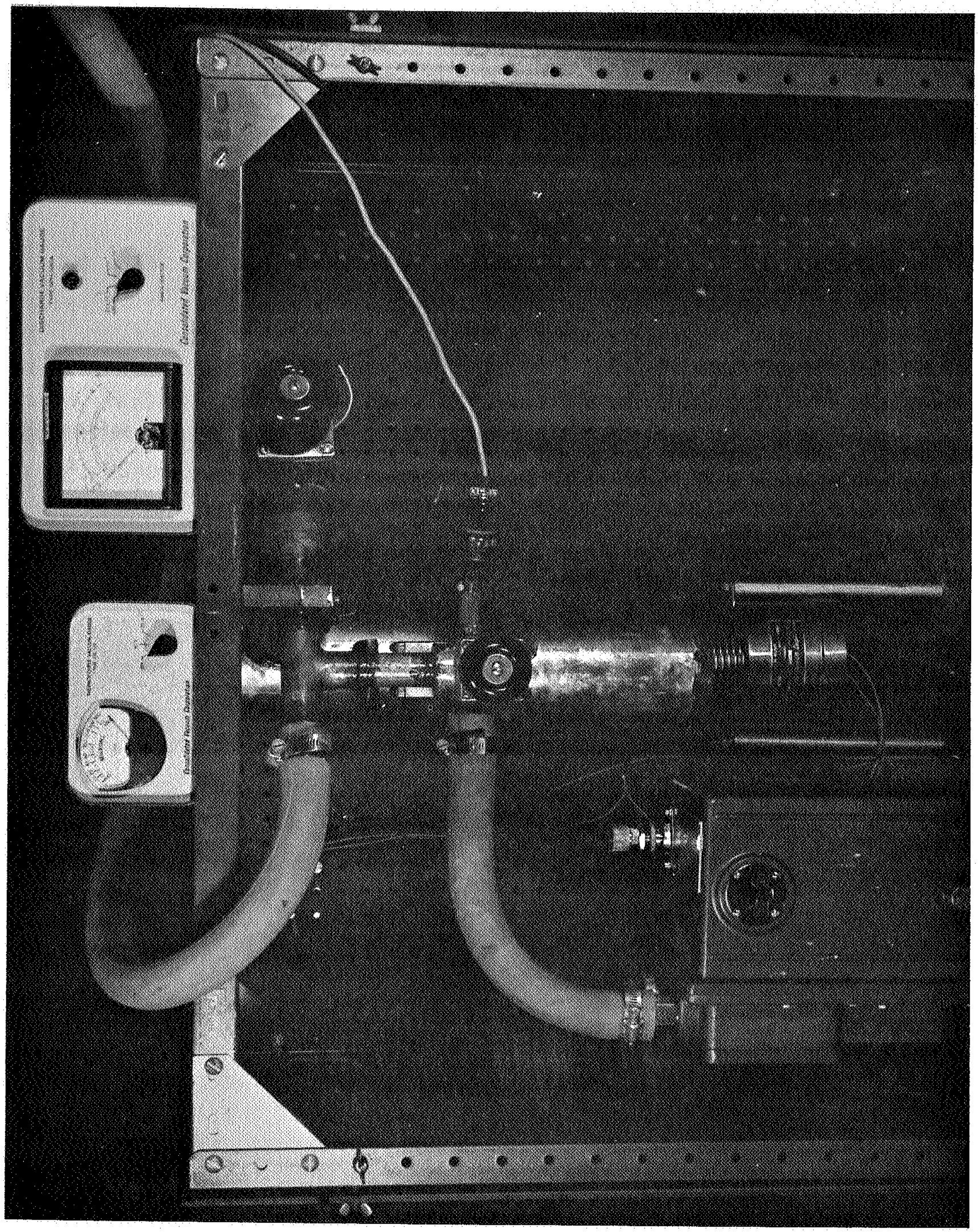

Figure 6. Vacuum System

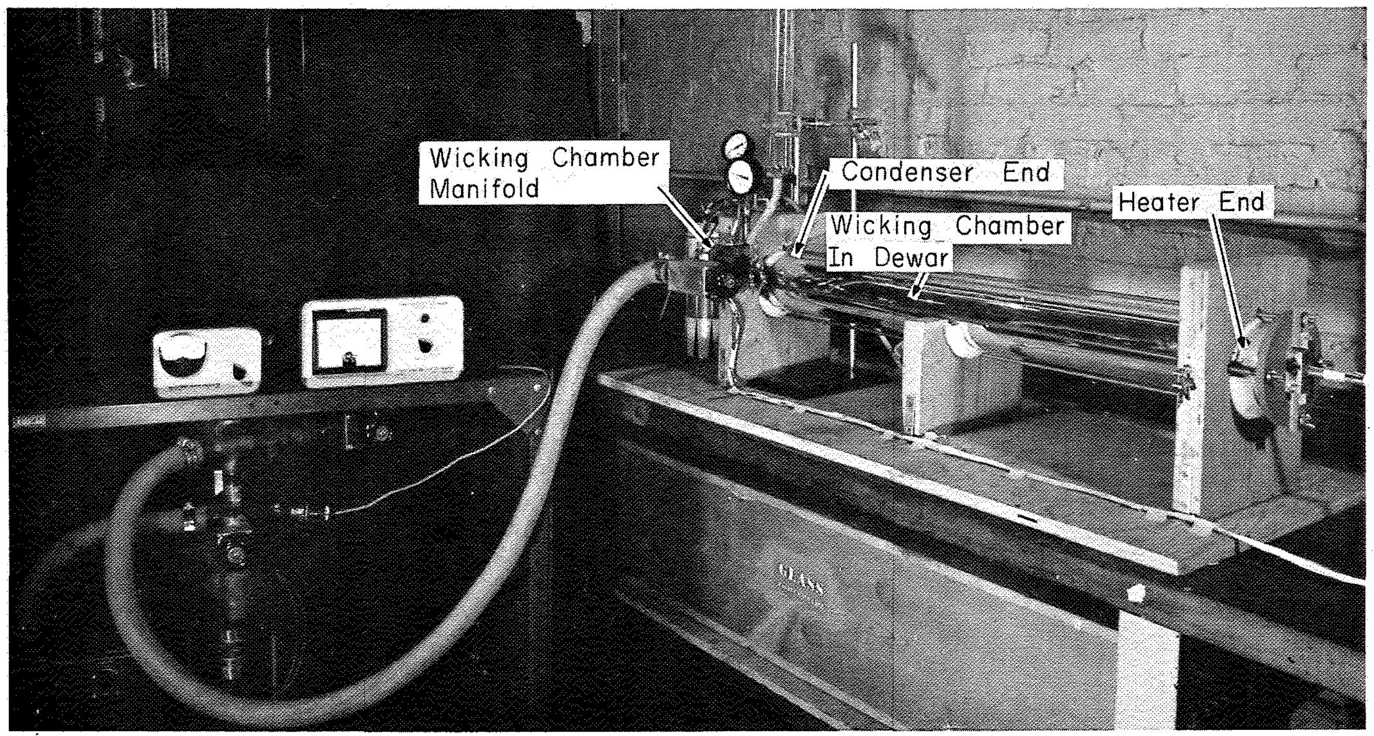

Figure 7. Heat Pipe and Vacuum System

all forty-five points for a period of time. A potentiometer was used to continuously monitor the temperature gradient between the main and guard heaters so that the guard heater variable transformer could be adjusted to minimize the heat loss from the main heater.

4.2 INITIAL TESTING PROCEDURE

The wicking chamber was evacuated to an initial absolute pressure of 8 x 10^{-2} mm of Hg. However, under this first evacuation, the wicking cage collapsed by buckling the stainless steel rod columns. After sufficient strengthening of the cage, the chamber was evacuated to 15 x 10^{-2} mm of Hg. Six hundred milliliters of distilled water was then injected into the chamber. This large amount of water proved to be much more than was necessary.

During the first application of heat to the wicking chamber, no heat was transferred to the condenser end and a temperature distribution of 5 to 10°C was established. Much purging of the wicking chamber was required before the temperature in the chamber equalized and a temperature gradient across the condenser aluminum disk was established.* After establishing the heat flux, further purging was required at about fifteen minute intervals. Without this purging, the temperature distribution in the wicking chamber widened and the wicking material appeared to begin "burn out" at less than 10 watts input power. With the intermittent purging, the maximum heat transport rate was approximately 10 watts.

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*Purging of the wicking chamber was performed by cracking (opening very slightly) the large vacuum valve while the mechanical vacuum pump was running. This procedure drew out of the chamber the non-condensible gases which had formed a barrier between the aluminum disk and the newly evaporated water vapor. The valve was opened for 1 to 5 second durations.

During testing, heat inputs were varied from an initial zero watts to a maximum of 60 watts at 5 watt intervals. Each input wattage was maintained until all temperature in the system stabilized. After stabilization, the heat input was stepped up or down, depending upon the test being performed. Even though runs of greater than 10 watts were obtained, 10 watts was the maximum repeatable heat flux before "burn out" occurred.

Since the time constant of the heater assembly and, in particular, the heater aluminum disk temperature were large (approximately seven minutes for 63.2% response of steady state for the aluminum disk), the "burn out" wattage was, at first, difficult to determine experimentally. When a given heat input was maintained for at least 10 to 15 minutes, 10 watts was the maximum maintainable input wattage. At wattage inputs above 10 watts, the heater aluminum disk temperature continued to increase without reaching a steady state value. This fact would indicate that the circular section of Refrasil attached to the aluminum disk began to dry out. Then the aluminum disk temperature could continue to increase and begin to diverge over the surface of the disk.

4.3 NORMAL TEST RUN

After initial charging of the wicking chamber, the heat pipe continues to operate until either non-condensible gases collect at the condenser surface or both the heater and condenser ends are at the same temperature. If a temperature difference exists and the heat pipe is not operating, the wicking chamber must be purged lightly. The heat pipe should be allowed to operate without adding heat until both ends are at the same temperature. Then the main heater should be set at 2 or 5 watts, depending upon the size input change utilized. The temperature difference between the main and guard heaters should be monitored continuously. Then the guard heater is adjusted so that the temperature difference is approximately zero. The temperature difference should be monitored and the guard heater adjusted throughout the entire test run.

When the heater aluminum disk temperature stabilizes, the main heater input should be increased by the 2 or 5 watt change required. The guard heater wattage should also be increased accordingly to minimize the main to guard heater temperature difference. The variance of the wicking chamber temperatures should be observed continuously. When the variance exceeds $\frac{1}{2}$°C, the chamber should be lightly purged. If the temperatures do not equalize within 20 seconds after the purge, the main heater should no longer be increased, for "burn out" is impending. At the same time, the temperature of the heater aluminum disk on the wicking chamber side should be observed. If these five temperatures on the disk continue to rise without reaching a steady state value or start diverging from each other, "burn out" could be impending. At "burn out" these temperatures will continue to diverge from each other regardless of the amount of purging. It is to be noted that the "burn out" can be a very gradual process.

Step down runs of the system were not performed successfully. The heat retention and time constants of the heater assembly and aluminum disk were too large.

After the wick "burn out", the heaters should be turned off. The wick must be allowed to resaturate before additional tests can be run.

4.4 WICKING MATERIAL TESTS

The wicking material, Refrasil #C100-28, used in the heat pipe was performance tested outside the heat pipe. Figure 8 shows the test apparatus used in these tests. The two objectives of the tests were to obtain fluid front displacement and volume input as functions of time data. Then the volume transport rate of the wick for any length was calculated in two ways. First, the transport rate was found using the equation

$$\dot{v} = AV \tag{12}$$

where $\dot{v}$ is the volume transport rate, A is the measured cross sectional area of the wick, and V is the velocity of the wetting front. The V term was determined by differentiating the displacement-time equation determined from Fig. 9. Second, the volume transport rate was determined by summing the total volume input to the wick over a given length. When this total volume was divided by the total elapsed time, the volume flow rate for a wick of one-half the total wick length used was determined. Equation (8) shows that the mass transport rate is inversely proportional to the total wick length. The volume rate for the length of wick desired can be scaled by multiplying by the appropriate ratio. This procedure for scaling the volume transport rates for different wick lengths was applicable only to the horizontal wick configurations, as can be seen from Eq. (6).

The wicking test apparatus, Fig. 8, consisted of three main elements. First, the distilled water was supplied by a 100 ml burette; all excess water dripping from the saturated end of the wick was

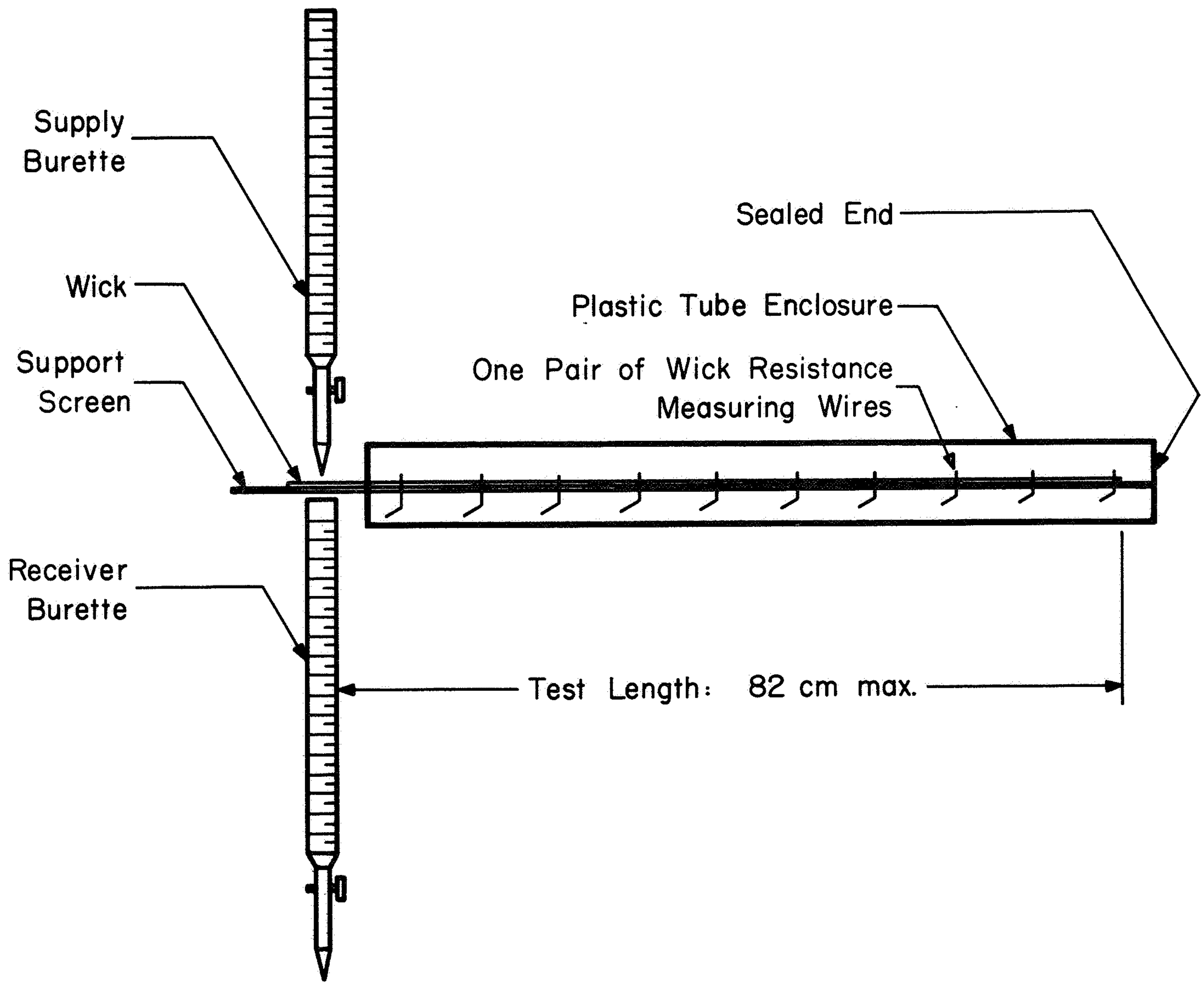

Figure 8. Schematic of Test Arrangement for Measurement of Capillary Flow in Horizontal Wicks

collected in a 50 ml receiver burette. The total volume transferred down the wick at any time was the difference between the readings of the supply and receiver burettes. Second, the wick was supported on a 1 cm-square mesh screen; a 3.2 cm inside diameter plexiglass tube was used to house the entire wick, except for the area where the water was introduced. This tube minimized losses from the wick by evaporation. Before each test, a saturated wick was placed inside the tube for several hours in order to saturate the atmosphere there. Third, pairs of copper wires were attached to the support screen at every 5 cm distance down the length of the wick. The insulation was removed from the ends of all the wires and the ends were then inserted into pores in the wick. The wires in each pair were separated by approximately 1 cm in a given cross section of the wick and each pair of wires was separated by a 5 cm distance down the wick. All the wires were connected to a 25-point thermocouple switch; the main connection of the switch was connected to an ohmmeter. When the wick was dry, the ohmmeter indicated an infinite resistance for any pair of wires. Immediately after the pores of the wick occupied by a pair of wires in any cross section were wetted, the resistance between wires changed (within approximately 2 sec) from an infinite to approximately 1MΩ resistance. When one cross section was wetted, the thermocouple switch was advanced to the next pair of wires to detect wetting of the wick at the next 5 cm interval. When the wetting front reached each pair of wires, all readings were taken. The readings included: supply and receiver burette volumes, total distance of wetting front down the wick, and total elapsed time of the test measured using a stop watch. From the distance and time data, a graph (Fig. 9) was constructed.

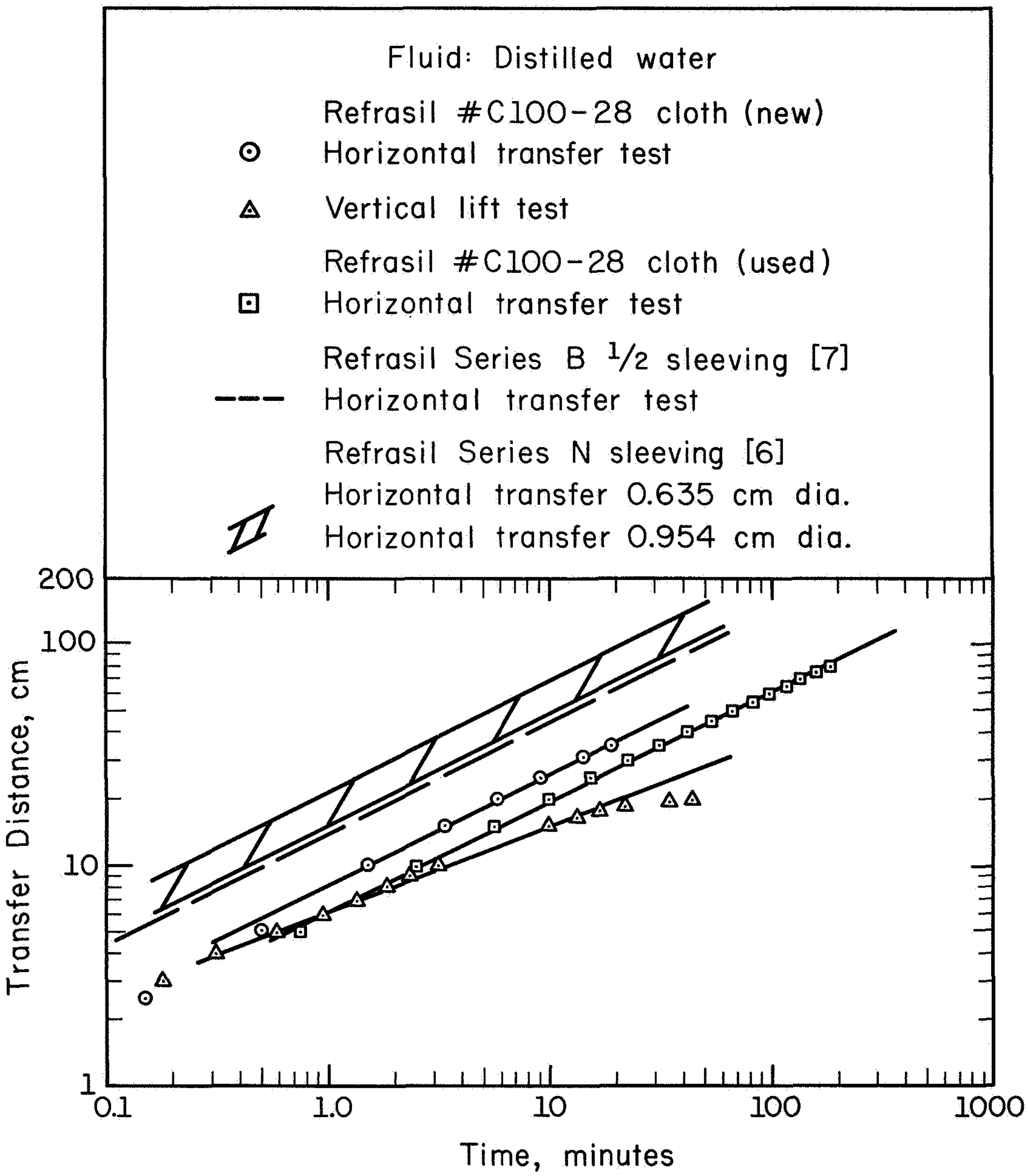

Figure 9. Wick Performance Graph

Maximum test wick width and length were 2.5 cm and 95 cm, respectively, for the test apparatus constructed.

5. DISCUSSION

For a "burn out" heat flux of 10 watts, which for the Refrasil #C100-28 used corresponds to 0.594 watts per cm width of wick, the volume transfer rate of water in the Refrasil wicking material required was

$$\dot{v} = \frac{Q}{h_{fg}\rho} = 0.247 \text{ cm}^3/\text{min} \ , \tag{9}$$

where Q equals the "burn out" wattage transferred and h_{fg} is the enthalpy of evaporation of water at 26.7°C (80°F). With a wick cross sectional area of 0.685 cm^2, the volume transfer rate per unit area becomes 0.361 $cm^3/min\text{-}cm^2$. The wicking chamber temperatures for all tests were within 5°C of the 26.7°C temperature used for the calculations.

From the wick performance tests on the Refrasil #C100-28, the volume flow rates per area were 0.299 to 0.424 $cm^3/min\text{-}cm^2$. The lower value was determined from the volume input measurements, whereas the upper value came from velocity calculations on the test data (Fig. 8). This wicking material was unused.

Another wick performance test was conducted on the actual wick used in the heat pipe. The results indicated approximately 45% less than the volume transport rates per cm^2 area for the unused material; the values were 0.163 to 0.229 $cm^3/min\text{-}cm^2$ for volume and velocity calculated, respectively. This decrease in wicking capability could be due primarily to wick contamination during the four months while it was used in the heat pipe. The used wick had acquired a yellow-brown appearance in contrast to the white color of the unused wick.

References [6] and [7] give water flow rates through horizontal wicks for different specimens of Refrasil. When scaled to the 81.9 cm length used in this heat pipe, the volume transfer rates per cm^2 area from [6] were 1.39 and 2.82 $cm^3/min\text{-}cm^2$ for the 0.954 and 0.635 cm diameter sleeving, respectively. These values are from 365% to 567% greater than the wicking tests performed by the author. The results of [7] lie between the data of [6] and those obtained in this work. There are three main reasons for large variations in liquid transfer rates. First, the Refrasil used in this experiment probably had a different pore size than the samples used by the other authors. As shown in Eqs. (6) and (7), the pore radius is one of the independent variables for the mass or volume flow rates. Second, the sleevings were flattened for the horizontal tests in the previous references. The interaction of the touching surfaces of the inside diameter may have caused errors in the transport rate. Third, in [6] the water was removed from the end of the wick by adding heat from a gas burner. This addition of heat created a large temperature rise and a resulting possible 20% change in the surface tension of the water. Since surface tension is the main pumping pressure, the temperature rise could cause error in the transfer rate. Figure 8 shows the comparison among the wicking materials.

The experimental "burn out" transfer rate per cm^2 area (at 10 watts) of 0.361 $cm^3/min\text{-}cm^2$ lies well within the range predicted by the author's horizontal wick performance tests. Actually the experimental value coincides with the average of the predicted rates for the unused Refrasil #C100-28 and within 36% of the upper predicted value for the used Refrasil #C100-28.

The maximum vapor velocity in the wicking chamber at "burn out" was approximately 446 cm/min. This value was far below the speed of sound in water vapor, 2.60×10^6 cm/min at 26.7°C. The corresponding Reynolds number at "burn out" was 14.5, indicating laminar flow.

If the entire cross sectional area of the wick were capable of transporting liquid, the water velocity in the wick while approaching "burn out" would be approximately 0.361 cm/min. Actually much of the area is not capable of transporting liquid due to the presence of the fibers of the Refrasil. The mean velocity of the liquid in the wick probably would be approximately 2 to 5 times that value stated above. A maximum value of 1.805 cm/min seems reasonable. The corresponding maximum Reynolds number was 56.3 in the wick with water at 26.7°C.

Usable data for the heat flux through the aluminum disks were not obtained. Thermocouple points had been attached to the aluminum disks with epoxy for the purpose of obtaining both absolute temperature readings on the disks and relative temperatures across the disks. This relative temperature, along with the accurately determined conductivity of the aluminum disks, was to be used for calculating the heat flux. However, a thin layer of epoxy had been applied to the bare thermocouple wires before attaching them to the disks in an effort to prevent electrical shorting of the thermocouple wires. The epoxy did indeed prevent electrical shorting, but also introduced a large thermal resistance. This resistance was large enough to cause errors in the differential temperature of 50 to 1500%. The maximum differential temperature times the disk conductivity yielded a heat flux of up to 15 times greater than the heat applied through the main heater. This measurement was obviously in error.

With the system used, no transient temperature lag from one end of the wicking chamber to the other end was observed during heat input changes. Apparently the time constants of the heat input changes were much larger than the temperature equalizing time constant of the wicking chamber. The time constant for the temperature on the wicking chamber side of the heater aluminum disk was approximately seven minutes for a 63.2% response of steady state.

6. CONCLUSIONS

The heat pipe construction was completed and initial experimental operating data were obtained. The maximum heat transport rate was 10 watts. For the Refrasil #C100-28 used, the 10 watt "burn out" point corresponds to 0.594 watts per cm width of the wick. This value was well within the power level of 8.27 to 11.8 watts predicted from horizontal wicking tests performed on the Refrasil #C100-28. Both vapor and liquid flows in the wicking chamber were found to be laminar.

Usable temperature drops across the aluminum disks were not obtained; temperature variances within the wicking chamber were found to be consistently less than $\frac{1}{2}^{o}C$ over the entire 81.9 cm wick length. With the system used, no transient temperature lag from one end of the wicking chamber to the other end was observed during heat input changes. Apparently the time constants of the heat input changes were much larger than the temperature equalizing time constant of the wicking chamber.

During this initial testing, many problems were encountered. Most of them were solved during the test runs and the main operating parameters of the heat pipe system were determined. Perhaps the most important fact of the initial testing was the resulting closeness of the experimental "burn out" to that predicted from horizontal wicking tests performed by the author.

7. RECOMMENDATIONS

7.1 HEAT PIPE MODIFICATIONS

1. A new wicking cage should be made. Narrow supports for locating the wicking material should be provided at a 10 to 12 cm spacing. If the cage is made solid (nonadjustable), attachment of the wicking material may be facilitated greatly.

2. Since high temperature build up was encountered around the heater pack, several of the mica sheets and the copper disk should be removed from between the main heater and the heater aluminum disk. With less thermal insulation, a greater heat flux could be realized with less temperature differential.

3. Special thermocouple wires should be used. These wires should have measuring points coated with material having high electrical and low thermal resistances. The special wires could be placed on each side of the aluminum disks to accurately measure the differential temperature drops. With these temperature measurements, accurate values for the axial heat flux could be calculated.

4. A small needle valve should be inserted on the manifold between the vacuum system and the wicking chamber. This valve would provide fine control for purging non-condensibles from the wicking chamber.

5. During reassembly of the entire heat pipe, special care should be taken to double check all joints for possible leaks. After assembly of the heat pipe, all joints should be coated with several thin layers of "Glyptol" or its equivalent.

6. The wicking chamber of the heat pipe should be evacuated to about 10^{-2} to 10^{-3} mm of Hg. before charging the chamber with water.

When evacuated, the chamber should be charged with approximately 100 to 150 milliliters of distilled water. Over 150 milliliters of water may result in an excessively large pool of liquid on the bottom of the wicking chamber. If the heat pipe length is changed, the amount of charge should be changed proportionately.

7.2 WICKING MATERIAL TESTING

Wicking material testing experiments should be continued to determine the volume transport qualities of different wicking materials. The experiment should be carried out under constant temperature conditions. A burner should not be used at the evaporator end of the wick, since the elevated temperature may change the surface tension and viscosity of the working fluid. If obtained with heating, the data may not represent the true operating conditions inside the heat pipe.

7.3 FUTURE HEAT PIPE TESTS

1. Test runs should be performed on shorter lengths of the wicking chamber to determine what effect the wick length has on the "burn out" wattage. For the horizontal case Eq. (8) should be further verified.

2. Fluids other than water should be employed in the wicking chamber to determine their operating characteristics. These fluids should then be compared to water for possible future application.

3. Combinations of two different fluids should be employed in the wicking chamber. Compatibility and other operating characteristics should be determined for all *practical* combinations of the fluids.

4. A new heat input unit with a sufficiently small time constant, so that the transient phenomena inside the wicking chamber may be observed during heat input changes, should be designed and built.

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LIST OF SYMBOLS

A	cross-sectional area of wick (ℓ^2)*
A_c	minimum cross-sectional area of wicking chamber (ℓ^2)
C	arbitrary constant (M-ℓ/t)
g	acceleration of gravity (ℓ/t^2)
h_{fg}	latent heat of vaporization of liquid (q/M)
K	wick permeability (ℓ^2)
L	actual length of wicking material (ℓ)
$\dot{m}$	mass flow rate (M/t)
ΔP_c	capillary pumping head (F/ℓ^2)
ΔP_g	gravitational head (F/ℓ^2)
ΔP_L	liquid viscous drag (F/ℓ^2)
ΔP_v	vapor pressure drop (F/ℓ^2)
Q	heat transfer rate (q/t)
r	mean pore radius of wicking material (ℓ)
r_c	effective pore radius of wick at condenser (ℓ)
r_e	effective pore radius of wick at evaporator (ℓ)
V	velocity (ℓ/t)
$\dot{v}$	volume flow rate (ℓ^3/t)
w	total wick width (ℓ)
θ_c	liquid contact angle in wick at condenser
θ_e	liquid contact angle in wick at evaporator
μ	absolute viscosity of liquid (F-t/ℓ^2) or (M/t-ℓ)
ν	kinematic viscosity of liquid or vapor (ℓ^2/t)
ρ	liquid density (M/ℓ^3)

*Dimensions in parentheses are: F - force, M - mass, ℓ - length, q - heat (F-ℓ), t - time.

σ liquid surface tension (F/ℓ)

ϕ angle between heat pipe axis and gravitational field

APPENDIX A: SAMPLE CALCULATIONS

"Burn out" power rating per cm width of wick (Refrasil #C100-28):

$$\frac{Q}{w} = \frac{10 \text{ watts}}{16.85 \text{ cm}} = 0.594 \text{ watts/cm}$$

Required transport rate of water in the Refrasil for a "burn out" wattage of 10 watts:

$$Q = \rho \dot{v} h_{fg}$$

$$\dot{v} = \frac{Q}{h_{fg}\rho}$$

Q = 10 watts = 143.5 cal/min
h_{fg} = 582 cal/g @ 26.7°C.
$\dot{v}$ = volume transport rate
ρ = 1 g/cm^3 @ 26.7°C. for water

$$\dot{v} = \frac{143.5 \text{ cal/min}}{(582 \text{ cal/g})(1 \text{ g/cm}^3)}$$

$$\dot{v} = 0.247 \text{ cm}^3/\text{min}$$

Cross-sectional area of wick:

$$A = (\text{thickness})(\text{width})$$

thickness = 0.016 in. = $(4.06)(10^{-2})$ cm

width = 6.625 in. = 16.85 cm

$$A = (4.06)(10^{-2})(16.85) = 0.685 \text{ cm}^2$$

Required volume transport rate per cm^2 cross-sectional area:

$$\frac{\dot{v}}{A} = \frac{0.247\ cm^3/min}{0.685\ cm^2} = 0.361\ cm^3/min\text{-}cm^2$$

Available volume transport rates from horizontal wicking performance tests:

Wicking material: Refrasil #C100-28
Wicking fluid: distilled water
Fluid temperature: 26.7°C.

<u>Unused material; 2.50 cm wide, 31.0 cm long</u>

1. Calculated from volume input measurements:

$$\dot{v}_{(31.0/2\ cm)} = \frac{\text{total volume}}{\text{total time}}$$

$$= \frac{2.30\ cm^3}{14.33\ min}$$

$$= 0.1605\ cm^3/min$$

$$\frac{\dot{v}}{A}\ (15.5\ cm) = \frac{0.1605\ cm^3/min}{(2.50\ cm)\ (4.06)(10^{-2})\ cm}$$

$$= 1.58\ cm^3/min\text{-}cm^2$$

$$\frac{\dot{v}}{A}\ (81.9\ cm) = \frac{(15.5\ cm)(1.58\ cm^3/min\text{-}cm^2)}{(81.9\ cm)}$$

$$= 0.299\ cm^3/min\text{-}cm^2$$

2. Calculated from velocity which was derived from displacement vs time plot:

$$\frac{\log x}{\log kt} = \frac{1}{2}\ ,\ kt = x^2$$

$$k = \frac{(25.0)^2}{9.00} = 69.4 \text{ cm}^2/\text{min (using experimental point)}$$

$$69.4t = x^2$$

$$V = \frac{dx}{dt} = 34.7 \; \frac{1}{x} \; \text{cm/min}$$

$$V = \frac{34.7}{81.9} = 0.424 \text{ cm/min}$$

$$\frac{\dot{v}}{A} = V = 0.424 \text{ cm/min} = 0.424 \text{ cm}^3/\text{min-cm}^2$$

$$= 0.424 \text{ cm}^3/\text{min-cm}^2 \text{ for } 81.9 \text{ cm wick length}$$

<u>Used material; 2.5 cm wide, 80.0 cm long</u>

1. Calculated from volume input measurements:

$$\dot{v}_{(40.0 \text{ cm})} = \frac{\text{total volume}}{\text{total time}}$$

$$= \frac{6.40 \text{ cm}^3}{188.5 \text{ min}}$$

$$= 3.39 \times 10^{-2} \text{ cm}^3/\text{min}$$

$$\frac{\dot{v}}{A} \; (40.0 \text{ cm}) = \frac{3.39 \times 10^{-2} \text{ cm}^3/\text{ min}}{(2.50 \text{ cm})(4.06 \times 10^{-2} \text{ cm})}$$

$$= 0.334 \text{ cm}^3/\text{min-cm}^2$$

$$\frac{\dot{v}}{A} \; (81.9 \text{ cm}) = \frac{(40.0 \text{ cm})(0.334 \text{ cm}^3/\text{min-cm}^2)}{(81.9 \text{ cm})}$$

$$= 0.163 \text{ cm}^3/\text{min-cm}^2$$

2. Calculated from velocity which was derived from displacement vs time plot:

$$\frac{\log x}{\log kt} = \frac{1}{2}, \quad kt = x^2$$

$$k = \frac{(60.0)^2}{96.0} = 37.6 \ cm^2/min$$

$$37.6t = x^2$$

$$V = \frac{dx}{dt} = 18.8 \ \frac{1}{x} \ cm/min$$

$$\frac{\dot{v}}{A} = V = \frac{18.8}{81.9} = 0.229 \ cm/min$$

$$= 0.229 \ cm^3/min\text{-}cm^2 \text{ for } 81.9 \text{ cm wick length}$$

Liquid velocity in wick as acquired from the required volume transport rate, also assuming that entire wick area was capable of transporting liquid:

$$V = \frac{\dot{v}}{A} = 0.361 \ cm^3/min\text{-}cm^2$$

$$= 0.361 \ cm/min$$

If actual wick area for transporting liquid was approximately 1/5 total area, A, due to the Refrasil fibers:

$$V_{(max)} = 5V = 5(0.361 \ cm/min)$$

$$= 1.805 \ cm/min$$

Maximum Reynolds number of liquid flow in wick based on L_{max}:

$$Re_{max} = \frac{L_{max}}{\nu} \qquad \begin{aligned} L_{max} &= \text{assumed to be total wick width} \\ &= 16.85 \text{ cm} \\ \nu &= 0.540 \text{ cm}^2/\text{min at } 26.7^\circ\text{C for liquid} \end{aligned}$$

$$= \frac{(16.85 \text{ cm})(1.805 \text{ cm/min})}{(0.540 \text{ cm}^2/\text{min})}$$

$$= 56.3 \text{ for water in wick at } 26.7^\circ\text{C.}$$

Maximum vapor velocity in the wicking chamber as acquired from the required volume transport rate:

$$V_{max} = \frac{\dot{v}_v}{A_c}$$

$$\dot{v}_v = \dot{v}\,\frac{\text{specific volume of vapor}}{\text{specific volume of fluid}}$$

$$V_{max} = \frac{\dot{v}\,(\text{specific volume})_{vapor}}{A_c\,(\text{specific volume})_{fluid}}$$

$$\begin{aligned} \dot{v}_v &= \text{vapor volume transport rate} \\ \dot{v} &= \text{fluid volume transport rate} \\ &= 0.247 \text{ cm}^3/\text{min} \\ A_c &= 21.8 \text{ cm}^2 \end{aligned}$$

$$V_{max} = \frac{(0.247 \text{ cm}^3/\text{min})(633.7)}{(21.8 \text{ cm}^2)(0.0161)} \text{ @ } 26.7^\circ\text{C.}$$

$$= 446 \text{ cm/min}$$

Reynolds number of vapor flow in the wicking chamber, based on L_{max}:

$$Re_{max} = \frac{L_{max}}{\nu}$$

$$\begin{aligned} L_{max} &= \text{dewar diameter} \\ &= 7.62 \text{ cm} \\ \nu &= 234 \text{ cm}^2/\text{min @ } 26.7^\circ\text{C for vapor} \end{aligned}$$

$$Re_{max} = \frac{(7.62\ cm)(446\ cm/min)}{(234\ cm\ /min)}$$

$$= 14.5 \text{ for water vapor @ } 26.7^{\circ}C.$$